An Oil-Free Compressor for Gifford McMahon Cold Heads

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ABSTRACT

Industrial Research Ltd (IRL) has developed a novel metallic-diaphragm pressure wave generator technology for large high-frequency (30-60 Hz) pulse tube cryocoolers. A diaphragm pressure wave generator, using a conventionally lubricated motor-crank drive, provides a clean pressure wave for a cryocooler. High frequency pulse tubes are at their most efficient when closely coupled to pressure wave generators. There are, however, many applications which require the physical separation of the compressor from the cold head such as a Gifford McMahon (GM) system. Furthermore, GM cryocoolers perform well at low temperatures due in part to their low frequency (~1Hz) operation. GM cryocoolers typically employ compressors related to air conditioning compressors which need filters to remove the compressor oil from the helium flow, thus increasing maintenance. This paper describes the development of an oil free compressor for GM cryocoolers fabricated by fitting a reed valve assembly to the outlet of a 200ml swept-volume diaphragm pressure wave generator. The system was designed to charge a high pressure tank to 16 bar drawing gas from a low pressure tank held at 5 bar. Helium mass flow rate was predicted to be 3.5 g/s. Results of testing the compressor are presented.

INTRODUCTION

Industrial Research Ltd (IRL) has developed a novel metallic-diaphragm pressure wave generator technology for large high-frequency (30-60 Hz) pulse tube cryocoolers[1]. The IRL diaphragm pressure wave generator (PWG) employs an opposed pair of metallic diaphragms driven by a conventionally lubricated motor-crank drive. The metal diaphragms separate the clean cryocooler gas from the lubricated drive mechanism. High frequency pulse tubes are at their most efficient when closely coupled to their PWGs. There are, however, many applications which require the physical separation of the compressor from the cold head such as offered by Gifford McMahon (GM) systems. GM cryocoolers are a mature and well established technology that uses a constant pressure gas supply and rotary valve to provide pressure fluctuations for the cold head, as opposed to Stirling type systems that use the pressure fluctuations directly from the compression piston. The pressure fluctuations in GM cryocoolers are slow speed, typically around 1 Hz, which contributes to good performance at low temperatures. There are two types of GM cryocooler, the traditional cooler as developed by Gifford and McMahon in the 1960's [2] that uses a moving displacer-regenerator in the cold head, and low frequency pulse tubes that utilize

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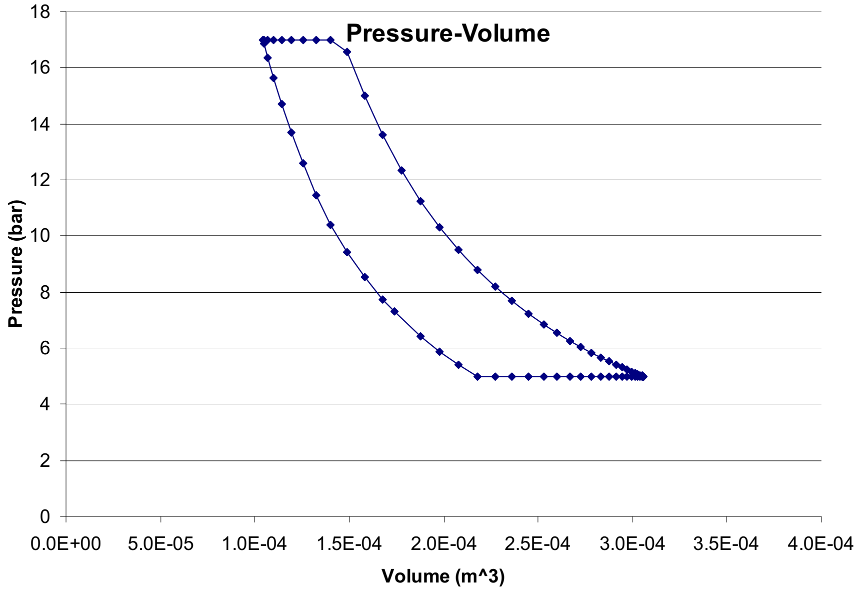

Figure 1. Predicted pressure-volume loop for 200ml swept volume compressor between 5 and 17 bar pressure reservoirs.

the traditional GM compressor and valves for pressure oscillation. The latter in particular are popular for liquid helium temperatures because of their long life and low vibrations.

GM cryocooler compressors are normally based on oil lubricated scroll compressors such as the Hitachi S600DH Helium Scroll compressor [3]. The compressed helium gas needs to be filtered to completely remove entrained lubricating oil before use as oil vapor can freeze in the cold head and reduce performance. Changing filter elements is a regular maintenance item. Helium scroll compressors have a limited operating temperature range, typically from 7 to 38°C. If the compressor is too cold then it will not start as the oil is too thick. If the compressor is too hot then excess oil vapor is carried into the outlet gas stream and significantly reduces filter element life. Helium scroll compressors are sensitive to pitch, and must be kept within 5 degrees of horizontal [4] so that excess oil will not become entrained in the gas flow. The requirement for level operation is not a hindrance in laboratory environments but limits application for mobile applications such as on board ships.

Rectification of the pressure wave from linear motor PWGs was successfully implemented and demonstrated by Q-Drive [5] whose test system delivered 3.6 g/s between 6.6 bar and 25 bar pressures for 7.5 kW power input. Q-drive also offer a 2S241C compressor quoted to deliver 2.1 g/s (800 std litres/minute) of Helium between 5 and 20 bar reservoirs for 4.1kW power input [6]. Table 1 summarizes a range of helium compressors for GM cryocoolers.

Table 1. Comparison of existing Helium Compressors for GM coolers.

Compressor	Inlet Pressure Bar	Outlet Pressure Bar	Mass Flow g/s	Power kW
Hitachi S600DH (compressor only)	5	17	3.8-4.5	4.4
Q-Drive	6	25	3.6	7.5
Cryomech CP950	5	21	3.1	5.5
Oxford Inst. M400			3.1	7

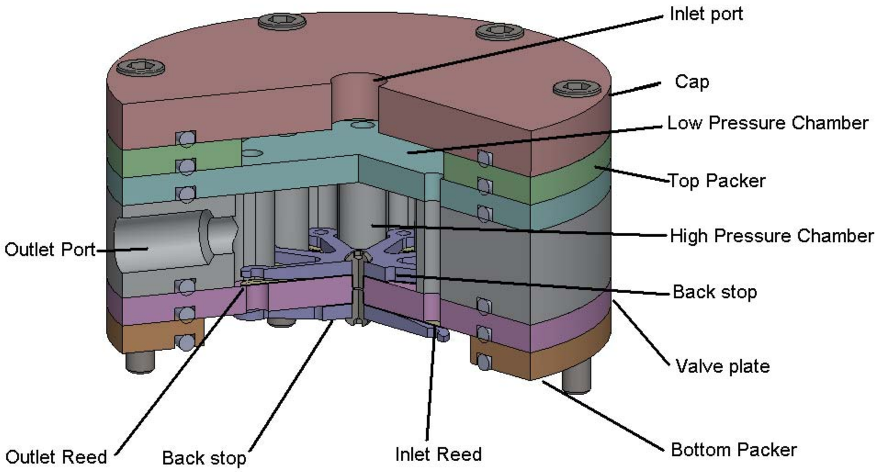

Figure 2. Design of the Reed valve system.

Metallic diaphragm PWGs have the potential to provide a useful alternative to linear motor or scroll based compressors for GM cryocoolers. Such a system would offer a compressor that does not require filtering of the helium supply, is more tolerant to temperature variation and inclination than a scroll compressor and at a cheaper price than a linear motor system. An additional feature that the diaphragm PWG can offer is variable frequency. The diaphragm system will work efficiently over a wide frequency range, from 10 to 60 Hz. Hence a variable pumping rate is available without compromising the overall pressure ratio. This allows a GM style cooler to vary its cooling power with load.

The current design of diaphragm PWGs provide up to 10 bar, peak to peak, pressure waves. This is due to a limitation in the load capacity of the bearings in the driving mechanisms, not the diaphragms. Modification of the drive system would be required to achieve the 12 to 16 bar pressure differential between high and low reservoirs as listed in Table 1. For the experimental work in this paper, the pressure differential was kept below 10 bar to avoid damage to the current prototypes.

The aim of this work is to determine the viability of IRL’s metallic diaphragm PWGs as compressors for GM type cold heads by demonstrating the ability to produce two to three grams per second mass flow of helium at approximately 10 bar pressure difference. Modeling, experiments, and development of a proof of concept prototype metallic diaphragm based compressor for GM cryocoolers are presented.

INITIAL MODEL AND PERFORMANCE PREDICTION

A simple spread sheet based model of a diaphragm compressor was constructed. The piston area, stroke and dead volume of the CHC200 PWG with 200ml of swept volume were used as inputs. It was assumed that the reed valves could be mounted directly in the compression space close to the diaphragm and that compression was adiabatic. Flow losses in valves or pipe work were not considered in this simple model. Figure 1 shows the predicted pressure-volume pumping curve. At 50 Hz cycle speed, the calculated mass flow rate was 3.5 g/s between reservoirs of 17 bar and 5 bar. Work done on the gas was 4.05 kW and 3.25 kW of heat needed to be rejected. With a PWG efficiency of 70%, 5.8 kW of electrical power would be required.

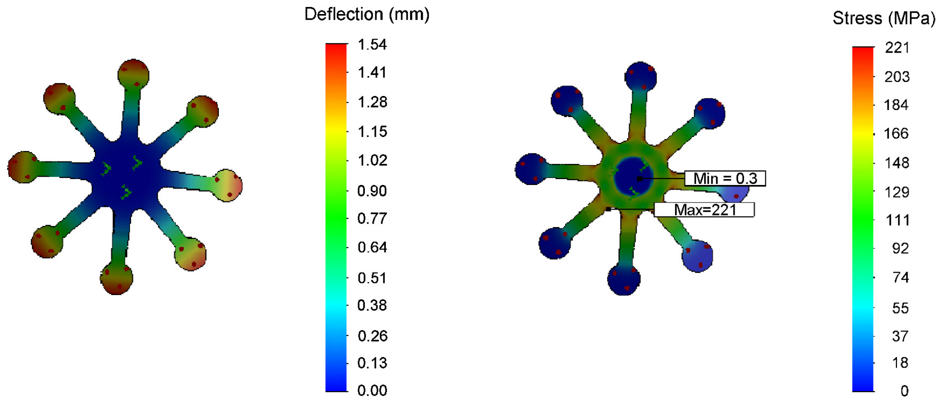

Figure 3. Finite Element analysis of reeds at maximum deflection allowed by the back stops.

This was close to the operating parameters of typical GM compressors such as those listed in Table 1, so it was decided to proceed with a proof of concept prototype test.

REED VALVE DESIGN

A critical part of the system is the valve design for rectification of the pressure wave. Reed valves were chosen for their proven long life and efficiency in many industrial piston compressors such as the Danfoss range of reciprocating refrigeration compressors [7].

Reed valves require a pressure differential to open the valve. This pressure differential was assumed to be the kinetic energy of the flow exiting the valves which is a function of the valve opening distance. The spring rate of the valve stem determined the opening distance and hence the pressure required to pass a given flow. The final reed arrangement, Figure 2, had eight valves each way in order to keep the flow velocities significantly sub-sonic.

The valve opening distance, and hence stress, was limited with back stops machined to the deflected shape. Maximum mass flow through the valve before the stops were reached was 40 g/s at a pressure drop of 0.2 bar.

Finite element analysis was used to calculate stresses and deflection of the reed under a minimal pressure drop. Deflection and pressure force were used to calculate the spring rate for valve opening and stress at maximum deflection was used to determine the shape of the backstop to limit reed movement. The material used for the reeds was Stainless Shim steel, 0.2 mm thick with a yield stress of approximately 800 MPa. The resonant frequency of the reed was calculated to be 337 Hz, which is high enough above the desired operating frequency of 50 Hz to allow the reeds to respond to the pressure pulses without a significant phase lag. The result is displayed in Figure 3.

EXPERIMENTAL SYSTEM DESIGN

An existing CHC240 PWG was used for testing. The CHC240 is a CHC200 with extended stroke, delivering 240 ml of swept volume. The PWG's top plate was not modified for the test which added 172 ml of extra dead volume to the system via the top plate outlet ports and its connection to the reed valve block. The total dead volume of the system thus increased to 360 ml. At this stroke and dead volume the simple model predicted that a pressure differential of 12 bar (17 to 5 bar) could not be attained, however a flow of 2.5 g/s was predicted at 50 Hz between 14 bar and 6 bar (8 bar pressure differential).

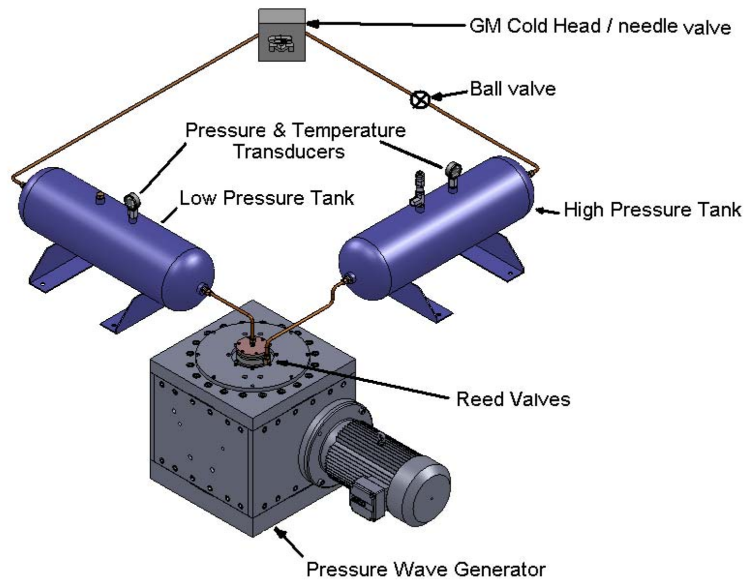

Figure 4. The experimental setup.

The reed valve block was bolted to the PWG top plate where the pulse tube would be otherwise connected, Figure 4. The outlet port of the valve bank was connected via a 1 m long x 3/8” diameter copper line to a 22.8 litre tank, the high pressure tank. The inlet port of the valve bank was similarly connected to another 22.8 litre tank, the low pressure tank. Pressure and temperature transducers were mounted on the high and low pressure tanks and in the PWG’s compression space. The diaphragm PWG’s gas spring was connected to the compression space via a long ¼” line and a needle valve to allow pressure equalization without disturbing the pressure wave.

A GM cold head was not available at the time of testing so its pressure drop was simulated by a needle valve connecting the high pressure and low pressure tanks. A ball valve in series with the needle valve was used for on-off switching of the flow between high and low pressure tanks.

EXPERIMENTS

Bare Volume Experiment

The first experiment involved compressing the dead volume of the system without the valve bank to determine the maximum possible pressure wave and compression characteristics of the system. Compression is described by the formula:

$$pressure\_ratio = \left(\frac{V_{\max}}{V_{\min}} \right)^n$$

where the polytropic exponent n = 1 for isothermal compression and 1.67 (γ) for adiabatic compression of Helium gas. Pure adiabatic compression was considered unlikely as the diaphragm system had a large area and the transfer port through the top plate consisted of 61 5 mm diameter holes 47 mm in length.

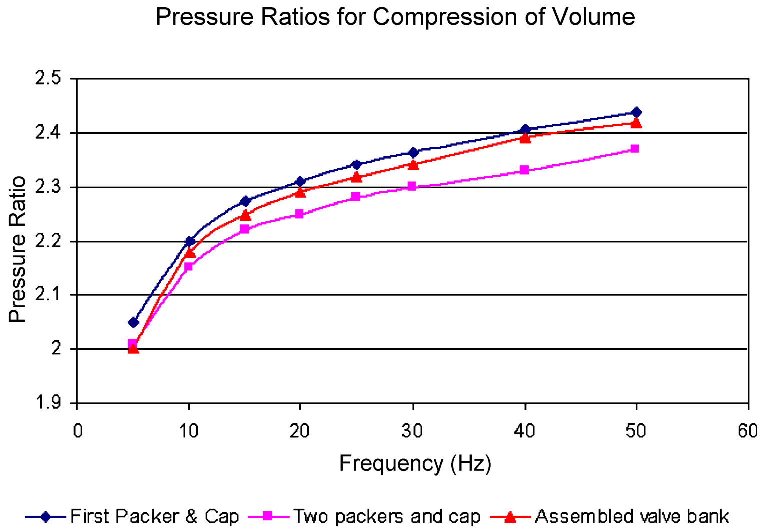

Figure 5. Pressure ratio for simple compression of volume.

The bare volume experiments were conducted at PWG frequencies from 5 Hz to 50 Hz and a charge pressure of 10 bar. Three cases were tested: The first was with the valve bank's bottom packer and cap, refer to Figure 2, with an average system volume of 405ml. The second was with an extra packer increasing the volume to 416ml and the third was with the fully assembled valve bank with its ports blocked off, average volume of 406ml. The swept volume for each case was 240ml. Figure 5 shows that the pressure ratio is heavily dependent on frequency, ranging from 2.0 to 2.4. The adiabatic pressure ratio for the first case was 2.76 and the isothermal pressure ratio was 1.84. Figure 5 shows the pressure ratios for the three cases at the range of speeds. At 50 Hz close to adiabatic compression was approached with an exponent n of 1.45, which produced the observed pressure ratio for all three cases. At 5 Hz the compression was trending towards isothermal with an n of 1.17. At 30 Hz an exponent n of 1.35 was produced. The diaphragm PWG has the capability of removing a significant amount of heat during compression as evidenced by its non-adiabatic polytropic compression exponent. Isothermal compression is considered a desirable feature for Stirling machines, however for compression past a reed valve it limits the ultimate pressure ratio that can be generated, and thus the mass flow through the reeds. Adiabatic compression, as opposed to near isothermal, allows the pressure in the cylinder space to climb faster hence opening the reed valves earlier and allowing more swept volume available for pumping gas.

Pump Characteristic Experiment

The original experimental method was to calibrate the needle valve between the high and low pressure tanks by discharging one tank into the other and deducing the mass flow from the change in mass of the tanks as calculated from the state equations using pressure, temperature and volume. After analysis of the first needle valve characteristic, at one quarter turn open, it became apparent that identical mass flow results could be deduced from initial charging of the high and low pressure tanks after starting the PWG from equilibrium with the ball valve closed. Producing a pump characteristic took a few seconds.

The final experimental procedure started with the whole system at its charge pressure. The ball valve was closed between the high and low pressure tanks. The data acquisition was set to record pressures and temperatures at a rate of 2 Hz. The PWG was started and the high pressure tank was charged from the low pressure tank. The mass of gas in each tank was calculated using

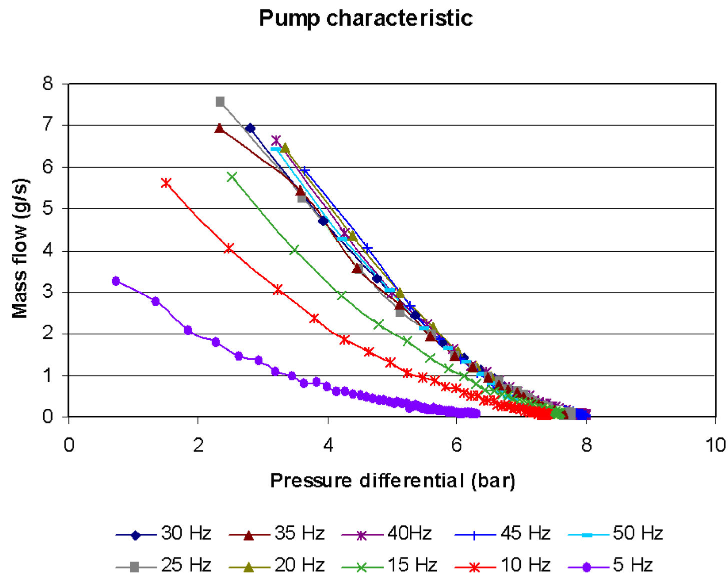

Figure 6. Pump characteristic for reed valve system mounted on the CHC240 PWG.

temperature and pressure. The change in mass of gas in each tank was the mass flow through the compressor at a given pressure differential. The system was then run at steady state conditions through the needle valve to verify the transient characteristic. The procedure was repeated for PWG speeds ranging from 5 to 50 Hz.

Figure 6 shows the pump characteristics for the transient experiments. A flow of 3 g/s was achieved at a 5 bar pressure differential for PWG speeds above 20 Hz. The pump characteristic remained unchanged for frequencies above 20 Hz. The pressure wave amplitude in the compression space was consistently around 8 bar (peak to peak) for speeds above 20 Hz which is consistent with the maximum pressure differential reached and the bare volume experiments. The frequency independence of pumping suggests a very effective low-pass frequency filter in the system. The steady state runs agreed with the transient pump characteristic. For the 20 Hz run, where the valve restriction was not important, 2 g/s flow was passed through the needle valve with a pressure difference of 6.0 bar, and 1.4 kW power input to the compressor. The system worked well at compressor speeds below 25 Hz, a speed at which the diaphragm PWG operates very quietly and efficiently. More efficiency could be gained at 25 Hz by the use of a 4-pole electric motor which runs at that speed. The output of the diaphragm PWG can be doubled, to achieve 3 g/s mass flow at 25 Hz, by fitting a second valve bank in place of the gas spring, on the bottom diaphragm. This has yet to be tried. The mechanism for the low pass filter was unclear so a more sophisticated time step model of the system was constructed using Matlab software.

Improved Time Step Model

A time step model of the transient experiment was constructed using Matlab. The time step model included the valve opening heights, pressure losses and the high and low pressure tanks. All the dead volumes in the experimental system were included. The frequency dependent polytropic exponent from the bare volume experiments was used.

The theoretical compressor characteristic is shown in Figure 7 for 10, 30 and 50 Hz and agrees well with the experimental results. The model was able to show the sharp low pass frequency filter effect observed in the experiments. The model indicated that the sharp frequency cut-off was due the pressure drop through the reed valves. The flow through the reed valves was

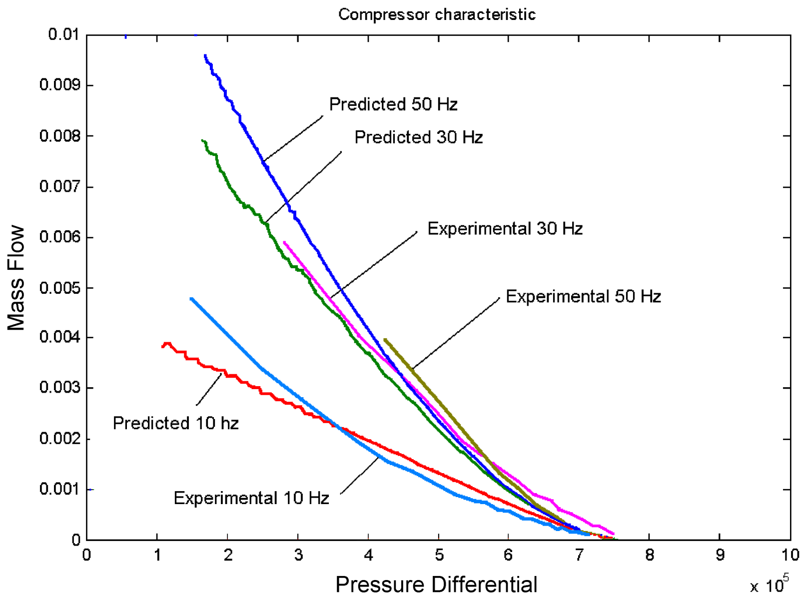

Figure 7. Comparison of the predicted and experimental pump characteristic for reed valve system mounted on the CHC240 pressure wave generator.

determined by the pressure differential between the compression chamber and tank. At high frequencies there was not enough time to push all the gas available from the piston stroke through the valve bank before the piston retracted and the valve closed. Over 20 Hz the flow through the reeds became a function of the average pressure differential across the valve during the valve opening time. As frequency increased, more cycles per second were experienced but the proportion of time the valve was open and the average pressure differential remained the same. The model showed a high sensitivity to the stiffness of the valve spring.

In the final DC compressor design, the dead volumes inherent in the current top plate and plenum on reed block would be removed. With the current reed valves mounted directly in the top plate, a flow rate of 3 g/s was predicted at a pressure differential of 7.5 bar between high and low pressure reservoirs. Opening up the reed ports to free the flow for high frequencies predicted a flow of 3 g/s at a pressure differential of 9 bar.

DISCUSSION AND CONCLUSIONS

A diaphragm based PWG was fitted with a set of reed valves to rectify its flow and charge high and low pressure reservoirs suitable for running a GM cryocooler. A sharp low pass filter effect was observed due to excessive flow restriction in the valves and transfer lines. The experimental pressure ratio was lower than desired due to extra dead volume incurred in mounting the valves on the top plate.

A Matlab time step model was produced that predicted the behavior of the system. The stiffness and associated pressure drop of the reed valve and delivery lines dominated the flow at high frequencies. With the reed system flowing freely and the compression space dead volume minimized, the model predicted a practical working mass flow of 3 g/s at a pressure differential of 9 bar, which is close to the compressors commonly available.

The next step for this work is to modify the reed valves to free up flow for high frequencies and reduce dead volume, the oil-free diaphragm compressor will then be ready for connection to a GM cold head to verify its operation as a cryocooler.

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